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STEADY-STATE PERFORMANCE OF A MODIFIED RANKINE CYCLE SYSTEM USING WATER

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SUMMARY

The Rankine cycle used for this investigation consisted of three fluid circuits. The heating loop supplied heat to the vapor loop boiler and the cooling loop removed the heat of condensation from the vapor loop condenser. In addition to the boiler and condenser, the vapor loop contained an unchoked vapor orifice and a circulating pump. The system was operated with fixed flow conditions in the heating and cooling loops and several values of vapor loop flow rate over a range of vapor loop fluid inventories. It was found that an increase of fluid inventory with a fixed vapor loop flow rate resulted in increases in condenser inlet pressure and decreases in the system heat load and vapor quality. The sensitivity of the condenser inlet pressure to inventory changes was found to be less at the higher vapor loop flow rates than at the lower flow rates investigated.

High vapor loop flow rates were also associated with small values of the mean temperature difference between the vapor and the condenser coolant and somewhat lower boiler exit vapor qualities.

INTRODUCTION

Future space missions may depend on the availability of electrical power for electric propulsion, communications, operation of scientific experiments, and for life-support systems. At present there is considerable interest in the use of reactor heated Rankine power cycles in which a turbine is used to drive an electric generator (ref. 1). In this cycle the turbine exhaust vapor must be condensed and the heat of condensation must be rejected by radiation in space.

Two types of heat rejection systems can be employed: one in which the vapor is condensed directly in the radiator tubes, and another in which the vapor is condensed in a convectively cooled heat exchanger (referred to as the condenser) that is cooled by circulating a liquid coolant to a radiator. Information is needed not only on component performance but also on the behavior of the complete power system. The large number of variables in either type of system makes it difficult to predict how the boiler and condenser interact to produce a steady-state operating condition to satisfy a particular set of independent system variables. The system variables are heating loop flow rate, heating loop temperature at the boiler inlet, vapor loop flow rate, vapor loop fluid inventory, cooling loop flow rate, and cooling loop temperature at the condenser inlet.

As part of an overall program at the Lewis Research Center concerned with Rankine space power systems, an experimental three loop system (heating, cooling, and vapor loops) using water as the working fluid was constructed to investigate the steady-state performance of a convectively cooled condenser system. The purpose of the system was to serve as a test bed for the study of two-phase system performance and as a pilot model for a similar alkali metal facility under construction at the Lewis Research Center.

The water system was composed of three fluid circuits: the heating loop, the cooling loop, and the vapor loop. The vapor loop contained a multitube boiler which could produce vapor at near atmospheric conditions, an unchoked vapor orifice between the boiler and the condenser, a shell and tube condenser, and a centrifugal pump to return the condensate to the boiler. Both the boiler and the condenser were mounted in a common horizontal plane to minimize the effects of gravity on the operation of the loop. An unchoked vapor orifice was selected to simulate the low pressure drop of the alkali metal facility. The boiler was heated with hot water to simulate a space system utilizing a liquid cooled reactor for the heat source. The condenser was liquid cooled to simulate a system with a liquid cooled condenser and a radiator as the heat sink. The operating pressure level of the facility was chosen such that the ratio of liquid density to vapor density of steam would be in the same range as that expected for an alkali metal system.

Data are presented to show the steady-state variation of the vapor loop pressures, temperatures, heat load, and vapor quality with changes in the vapor loop flow rate, vapor loop inventory, and coolant inlet temperature to the condenser. No attempt was made to study the transient or starting behavior of this system since work of that nature was being conducted under NASA contract (ref. 2).

APPARATUS

The water system was composed of three fluid circuits: the heating loop, the cooling loop, and the vapor loop. A schematic flow diagram of the system is presented in figure 1(a). The heating loop supplied heat to boil the water in the vapor loop and the cooling loop removed the heat of condensation and necessary condensate subcooling from the vapor loop. All three loops used water as the working fluid. The control and instrumentation locations are shown in figure 1(b). A photograph of the installation is presented in figure 2.

Heating Loop

The heating loop (fig. 1) was designed to allow independent control of both temperature level and flow rate of the heating water at the entrance to the boiler shell. Laboratory steam (100 psig max.) was utilized as the heat source. The water from the boiler shell was heated with steam by jet mixing in the heating tank whose capacity was 75 gallons. The tank was maintained at a constant pressure of 77 psia (310° F) by a steam pressure regulator. When the system heat load increased, because of a change in the demands of the vapor loop, the water entering the tank from the boiler shell was cooler and an increased quantity of steam was condensed in the tank. This, in turn, tended to lower the pressure in the tank. However, the steam pressure regulator maintained the tank pressure nearly constant by admitting more steam. The supply steam flow rate was thus made to be a function of the heat load requirements of the vapor loop for a given heating loop temperature (tank pressure) and flow rate. A liquid drain valve was used as required to expel condensed steam from the heating tank when the liquid level became excessive. A small cooler (fig. 1(b)) was installed between the heating tank and the boiler inlet to subcool the heating liquid by 10° F. This was necessary to prevent the possibility of vaporization of the heating liquid due to pressure losses at the entrance to the boiler shell. The flow rate in the heating loop was remotely adjusted with a throttle valve between the pump and the heating tank.

Instrumentation in the heating loop (fig. 1(b)) included the measurement of flow rate with a turbine-type flowmeter located at the pump discharge and the measurement of water temperatures at the inlet and outlet of the boiler shell with thermocouples. The latter were located at the center of a low velocity section of pipe near the entrance and exit of the boiler shell. In addition, the temperature drop of the heating liquid was measured directly with differential thermocouple probes.

Cooling Loop

The cooling loop was also designed to allow independent control of temperature and flow rate at the inlet to the condenser shell. In this loop hot water exiting from the condenser shell was mixed with cold domestic water in a cooling tank (fig. 1). Water at the desired temperature was then pumped from the cooling tank to the condenser shell. The flow rate of domestic water into the cooling tank was adjusted manually through a drain valve on the cooling tank. The pressure in the cooling tank was maintained nearly constant by the pressure regulators in the domestic water supply (fig. 1). The coolant flow rate to the condenser shell was manually controlled with a throttle valve between the pump and the condenser.

Instrumentation in the cooling loop (fig. 1(b)) included the measurement of the flow rate to the condenser shell with a turbine-type flowmeter, the measurement of temperatures at the inlet and outlet of the condenser shell with thermocouples, and the direct measurement of the coolant temperature rise across the shell with a differential thermocouple. The temperature probes were located near the center of the coolant stream as close to the inlet and outlet of the condenser shell as possible.

Vapor Loop

The vapor loop consisted of a shell and tube boiler, an unchoked orifice, a shell and tube condenser, a circulating pump, a flow control valve, and an inventory control tank. The boiler and condenser were installed in the same horizontal plane in order to minimize the effect of gravity on the operation of the loop. The vapor loop components are shown in figure 2. A description of the components of the loop is now presented.

Boiler. - The boiler size necessary to operate the facility was estimated from the boiling heat-transfer relations of reference 3. The conditions used for this estimate were a vapor loop flow rate of 500 pounds per hour, a saturation pressure of 20 psia, a heating loop flow rate of 8000 pounds per hour, and a heating loop boiler inlet temperature of 300° F. A commercially available boiler was purchased with more than the estimated surface area.

The boiler (fig. 3) was a single-pass shell and tube counter flow unit with 116 stainless-steel tubes each with a 0.375-inch inside diameter by 0.028-inch wall thickness and 5 feet long. Tubes were set on a 29/64-inch triangular pitch in a stainless-steel shell of 6-inch schedule 40 pipe. The shell was equipped with baffles set 3 inches apart to mix the heating fluid. The heating fluid flowed outside the tubes (shell side) and vapor was generated within the tubes (tube side). The tubes were connected at each end in plenum chambers to distribute the inlet liquid and to collect the vapor.

Vapor orifice. - A 1.14-inch-diameter orifice was installed in the 1.5-inch schedule 40 pipe connecting the boiler outlet to the condenser inlet to more closely simulate the flow restriction present in the alkali metal facility. The pressure drop between the boiler outlet and the condenser inlet was always less than that required for sonic flow in the orifice.

Condenser. - The condenser was designed using the condensing and liquid heat-transfer correlations of references 3 and 4. Small-inside-diameter tubes were selected to allow horizontal vapor flow without the severe slugging observed in larger tubes (ref. 5). The design conditions were a vapor flow rate of 500 pounds per hour, a saturation pressure of 20 psia, a vapor quality of 100 percent, a coolant flow rate of 6000 pounds per hour, and a coolant inlet temperature of 120° F.

The condenser shown schematically in figure 4 consisted of 19 stainless-steel tubes each approximately 10 feet long and each with a 0.3125-inch outside diameter and a 0.035-inch wall thickness. Vapor was distributed to the tubes from the inlet plenum and condensate collected in the outlet plenum. The coolant flow in the shell was directed opposite (counter current) to the vapor flow in the tubes. The shell was constructed with a concentric arrangement at the coolant outlet to provide uniform coolant flow as close to the vapor inlet as possible.

The condenser inlet total pressure was measured in the 1.5-inch pipe upstream of the inlet plenum with a strain-gage transducer (fig. 4). The condenser total pressure drop was measured directly with a strain-gage transducer connected to the vapor inlet and condensate outlet pipes. In addition to the inlet and outlet instrumentation, two condenser tubes (fig. 4) were instrumented to measure the axial temperature profiles in the condensing vapor. These thermocouples were inserted through the tube wall with the junction located near the center of the vapor stream. Typical temperature profiles are presented in figure 5 for a series of vapor flow rates. The interface location was taken as the point where the vapor temperature dropped suddenly. It was assumed that the average of the two interface locations was typical of all 19 tubes. The thermocouples were 8 inches apart and constructed with chromel-alumel wire swaged with magnesium oxide insulation inside a 0.040-inch-outside-diameter sheath. The sheathed thermocouples were bundled in the instrumentation channels and led out through the instrumentation ducts indicated on figure 4. The unused portions of the channels were filled with wire and soft solder to form a smooth inside shell diameter.

Condensate pump and flow control. - The condensate from the condenser was pumped back to the boiler with a sealed-rotor centrifugal pump (fig. 1). A bypass around the pump was included to ensure a sufficient flow to cool and lubricate it. The characteristics of this pump were such that little change in head rise occurred when the vapor loop flow rate was changed over the full range of the facility. The pump was operated at a fixed speed throughout this investigation and the flow rate in the vapor loop was controlled manually with a throttle valve installed between the pump and the boiler. The pressure drop across the throttle valve was large in comparison to the system pressure drop which may have contributed to the observed system stability.

PROCEDURE

Inventory Control

One of the independent variables of the system is the quantity of fluid present in the active volume of the vapor loop. The active volume of the vapor loop is defined as the total possible two-phase volume of the loop. This volume (shaded in fig. 6) included the boiler inlet liquid plenum, the boiler tubes, all of the vapor piping between the boiler and the condenser, the condenser vapor inlet plenum, and the entire volume of the condenser tubes. The volume of the entire vapor loop was found to be 1298 cubic inches using the method of appendix B. The volume of the all liquid portion of the loop was 225 cubic inches. Thus the active volume of the vapor loop was 1073 cubic inches. The quantity of liquid water contained within the active volume is defined as the active inventory and was expressed in cubic inches of water at 90° F. The active inventory was adjusted manually during operation by either pressurizing or evacuating the inventory control tank (fig. 1(a)) which was connected to the liquid portion of the loop between the condenser outlet and the condensate pump. An isolation valve was included between the tank and the loop to allow fixed inventory operation.

In order to provide a measure of the active inventory, the vapor loop was first filled with water and the liquid level in the inventory control tank was noted on the calibrated sight glass (fig. 1). During operation, the rise in liquid level in the inventory control tank was a direct measure of the quantity of liquid which had been displaced from the loop. The active inventory of the loop was then determined by simply subtracting the displaced volume from the measured active volume of the vapor loop.

Early operation of the facility indicated that two-phase operation of the vapor loop was confined to a very small range of active inventories and furthermore that this range was always at least $3\frac{1}{2}$ gallons less than the active loop volume. Consequently, an inventory control tank was designed with a 2-inch-diameter lower section in which the initial level could be accurately measured, a $3\frac{1}{2}$-gallon center section, and a 4-inch-diameter upper section in which the liquid level during operation could be accurately measured. The installation of the inventory control tank in the vapor loop is presented in figure 7.

Vapor Loop Fill and Degassing

The degassing procedure for the vapor loop involved the complete filling of the loop with distilled water and venting of the noncondensable gases. The vapor loop was initially evacuated to a pressure of about 700 microns of mercury and then allowed to fill with distilled water from a drum which was vented to atmosphere. Filling was terminated when the liquid reached a level in the inventory control tank which was higher than any point in the vapor loop. This provided an excess inventory and drove the noncondensables to the high points. The degassing problem was complicated by the horizontal attitude of the vapor loop and the multiplicity of narrow passages in the boiler and condenser. However, the loop did have high points at the boiler inlet, boiler outlet, and condenser inlet plenums where the gases tended to collect while the vapor loop liquid was circulating. Discharge of the gases from the loop was accomplished by manually opening vent valves which were installed at these locations and connected with transparent tubing to the laboratory exhaust system (5-in. Hg abs.).

After filling, the vapor loop was pressurized to approximately 20 psia, the pump was started, and the vapor loop flow rate set at the maximum value. The vent valves were opened and closed intermittently until no gas could be seen leaving the loop. To minimize the amount of dissolved gases, the vapor loop inventory was heated to approximately 200° F in the boiler by activating the heating loop. This temperature level was maintained for about 15 minutes. The loop pressure was then reduced to a value close to the vapor pressure and the loop vented again until no further gas flow could be seen in the transparent tubes.

Startup

After the filling and degassing process, any excess inventory in the vapor loop was removed through the fill valve (fig. 1). The liquid level in the inventory control tank was set at a point in the 2-inch-diameter section (fig. 7), and the vapor loop pressure and flow rate were adjusted to the desired value. The isolation valve between the vapor loop and the inventory control tank was left open throughout the starting sequence.

City water was supplied to the cooling loop through pressure regulators set to maintain a cooling tank pressure of 20 psig, and the cooling loop was vented at its highest points to expel any trapped air from the condenser shell. No effort was made to degas the cooling loop fluid because the effect of dissolved gases on the liquid heat-transfer coefficient was not believed to be severe in the temperature ranges encountered in the condenser shell. A cooling loop flow rate of about 6000 pounds per hour was established.

The water in the heating tank was heated to 310° F by gradually increasing the steam supply pressure. The air in the heating tank was removed through a vent located in the top of the tank. After the heating tank reached the desired pressure and temperature, the heating loop flow was initiated. As soon as the heating loop flow started, the pressures in the vapor loop and the temperature in the cooling loop were monitored on a continuous basis until steady-state operation was achieved. The conversion to two-phase operation caused a large, rapid increase in the vapor volume of the vapor loop. The displaced liquid from the vapor loop was allowed to enter the inventory control tank by manually venting the tank to maintain a nearly constant pressure. Venting was continued until a steady-state operating condition was reached at the desired condenser inlet pressure.

During the starting transient, the heat flow rate through the entire system increased rapidly. Consequently, the flow rate of city water to the cooling tank was increased to set the selected coolant temperature at the condenser inlet. Approximately 2 minutes were required to reach near steady-state operation.

Operation

A summary of the range of independent variables included in this investigation is presented in table I. The first group of data was obtained by holding the variables in the heating and cooling loops constant and then varying the vapor loop inventory to explore as wide a range of system operating conditions as possible for each of several values of vapor loop flow rate. The change of inventory was accomplished by changing the gas pressure on the inventory tank. This series of tests was performed to obtain the effects of both fluid inventory and vapor loop flow rate on the operation of the system.

In the next series of tests the effects of coolant inlet temperature were obtained with two fixed values of inventory over a range of vapor loop flow rate. In these tests, the isolation valve between the inventory control tank and the vapor loop was kept closed after the desired inventory was set.

The original and calculated data obtained during the investigation are given in table II. The methods of calculation used to reduce the raw data are presented in appendix C.

Data were recorded only when all variables had remained unchanged for a period of 15 minutes without readjustment of the controls. Approximately 3 minutes were required to record the data for each point. Temperatures were recorded on a self-balancing potentiometer. Pressures and some key temperatures and flow rates were recorded on a multichannel oscillograph. The condenser coolant temperature rise and the temperature drop of the heating liquid in the boiler were measured with differential thermocouples whose outputs were read manually on a precision potentiometer.

RESULTS AND DISCUSSION

The independent variables included in this investigation are the vapor loop flow rate, the vapor loop active inventory, and the condenser inlet coolant temperature. The effects of these variables on the steady-state performance parameters of the system (pressure, temperature, vapor quality, and heat load) are presented in the succeeding sections. The pressure and flow oscillations measured during steady-state operation will be presented first, however.

Steady-State System Stability

The system was operated in steady state over the range of conditions indicated in table I. As previously mentioned, some runs were obtained with variable inventory (the isolation valve open), and some runs were taken with fixed inventory (valve closed). It might be expected that the stability of the system would be greatly affected by the removal of such a large compressible volume as the inventory control tank. However, oscillograms of both modes of operation (fig. 8) indicate no significant change in the stability of the vapor loop flow rate and only slight changes in the character of the pressure traces. The magnitude of the oscillations of vapor loop flow rate and pressures are presented in table III as a percentage of the mean values for several cases of fixed and variable inventory operation. It can be seen that the spread of the data was about the same for both modes of operation. The very large percent oscillations listed for the condenser pressure drop in table III are believed to be the result of a condenser flow instability associated with vapor loop flow rates below about 250 pounds per hour. Under these conditions, the condensate in the condenser tubes may have formed small waves, or slugs, which became large enough to fill the inside tube diameter and trap a bubble of vapor. The trapped vapor then condensed rapidly causing a sharp pressure wave and a small flow disturbance to be transmitted throughout the system. A typical oscillogram of this type of instability is presented in figure 9. A similar characteristic was observed in the data of reference 5.

Effect of Vapor Loop Flow Rate

The effect of vapor loop flow rate on the temperatures and pressures of the fluids in the system is presented in figure 10 for fixed flow conditions in the heating and cooling loops and a vapor loop active inventory of 99 cubic inches. As the vapor loop flow rate increased, the heating loop boiler outlet temperature decreased (fig. 10(a)), which indicated an increase in the heat load of the system. The increased heat load is also reflected into the cooling loop where the condenser outlet temperature is observed to increase. At low vapor loop flow rates, the vapor loop condenser outlet (condensate) temperature closely approached the coolant inlet temperature. However, as vapor loop flow increased, the length of the condenser required for complete condensation increased, which resulted in a reduction of the length available for subcooling of the condensate. Thus the condensate outlet temperature increased. The increase in condensate temperature with vapor loop flow rate produced a similar increase in the boiler inlet temperature. The boiler inlet temperature was slightly higher than the condensate temperature because of the heat added by the vapor loop pump. Since the pump was operated at constant speed and the flow rate controlled with a throttle valve, the heat input of the pump was nearly constant and thus a higher temperature rise was observed at the lower vapor loop flow rates. Figure 10(a) also shows the temperature of the vapor at the boiler outlet and at the condenser inlet to be in close agreement with the saturation temperature at all but the lowest flow rates where the boiler could produce as much as 50° F of superheat.

The pressures at various points in the vapor loop are shown in figure 10(b) for the same conditions as discussed in figure 10(a). The pressure rise between the condenser outlet and the boiler inlet is supplied by the pump and throttle combination. It can be seen that the condenser inlet pressure increased nonlinearly with vapor loop flow rate with this fixed inventory and that the boiler inlet pressure was always 1 to 3 psi above it. Nearly all of the pressure loss between the boiler inlet and the condenser inlet is believed to have occurred at the vapor orifice because it was not possible to measure any pressure drop in the boiler.

The pressure at the condenser outlet is the result of the condenser inlet pressure and the condenser pressure loss, both of which increase with vapor loop flow rate. At a vapor loop flow rate of 350 pounds per hour the condenser outlet pressure reached a maximum value. Further increases in vapor loop flow rate produced lower condenser outlet pressures to accompany the higher condenser outlet temperatures seen on figure 10(a). The saturation pressure corresponding to the condenser outlet temperature is included on figure 10(b) to illustrate that the condensate flow actually approached a saturated condition at high vapor loop flows which eventually lead to cavitation of the vapor loop pump.

Effect of Active Inventory

As mentioned earlier, some data were obtained by varying the active inventory of the vapor loop at several values of vapor loop flow rate while holding constant inlet conditions in the heating and cooling loops (table I). The operating pressure, temperatures, heat load, vapor quality of the vapor loop, and the distribution of the active inventory were all found to be affected by variations of vapor loop flow rate and active inventory. The effects of flow rate and active inventory on the condenser and boiler inventories will be presented first.

The condensing length (a measure of condenser inventory) is presented in figure 11 as a function of active inventory for several vapor loop flow rates. It can be seen for a constant flow rate that the condensing length was reduced as more liquid was added to the vapor loop to increase the active inventory. Decreasing the vapor loop flow rate at fixed inventory tended to decrease the system heat load which also lowered the condensing length. If the volume of liquid contained in the vapor phase is neglected, the condenser inventory can be assumed to be equal to the volume of the liquid portion of the condenser. Condenser inventory is shown in figure 12 as a function of active inventory for several flow rates. Higher vapor loop flow rates correspond to lower condenser inventories. An increase in active inventory of 80 cubic inches resulted in an increase in condenser inventory of only 43 cubic inches at a flow rate of 314 pounds per hour. The remaining 37 cubic inches can be assumed to have gone into the boiler since the volume of liquid in the vapor phase of the entire vapor volume of the loop is only about 1 cubic inch. The boiler inventory was calculated by subtracting the condenser inventory from the active inventory and is presented in figure 13 as a function of active inventory for several vapor loop flow rates. The boiler inventory was found to increase with increases in active inventory as expected and the higher flow rates correspond to higher boiler inventories.

It would be expected that the pressure of the vapor would be affected by the changes in condenser and boiler inventories when the active inventory of the loop is changed. The variation of condenser inlet pressure with active inventory is presented in figure 14 for several vapor loop flow rates. It should be remembered that this system had no choked vapor orifice between the boiler and condenser. Thus the boiler outlet pressure was essentially the same as the condenser inlet pressure. As the inventory of the boiler and condenser were increased, by increasing active inventory, the pressure was observed to increase for all of the flow rates investigated. The slope of the curves, however, was observed to be less at the higher vapor flow rates.

The combined effects of vapor loop flow rate and active inventory on the condenser inlet pressure can best be seen by cross plotting the data of figure 14 to form a map of condenser inlet pressure against vapor loop flow rate for constant values of active inventory as shown in figure 15. It is not possible, from the data presently available, to present an exact explanation for the shape of these curves because the effects of the boiler and condenser heat transfer and pressure loss characteristics can not be separated. The interrelation of the boiler and condenser might be illustrated for a constant vapor loop flow rate as follows, however. When the condenser inventory is increased, by increasing active inventory, the temperature difference between the vapor and the condenser coolant must rise in order to accomplish the required heat rejection in the shorter condensing length (fig. 11). For a constant heat load (constant vapor loop flow rate), this change in temperature difference in the condenser can only be accomplished by increasing the condenser inlet pressure (saturation temperature) since the coolant inlet and outlet temperatures are fixed. The amount of the pressure rise depends on many factors such as the relation of system heat load and vapor quality to active inventory and the temperature difference between the vapor and the condenser coolant. If this temperature difference in the condenser is small, a large change in condensing length, or heat load, can be accommodated with a small change in pressure of the vapor. If the temperature difference is large, however, large changes in pressure will result from small changes in condensing length or heat load. This mean temperature difference in the condenser (T_{sat} - $T_{coolant\ average}$) is presented in figure 16 as a function of active inventory for several vapor loop flow rates. It can be seen that the condenser mean ΔT was lower, in general, for the higher vapor loop flow rates. Consequently, the observed convergence of the constant inventory lines of figure 15 at high vapor loop flow rates appears reasonable.

Because there was no choked orifice between the condenser and the boiler, any increase in vapor pressure at the condenser inlet, caused by an increase in inventory (fig. 14), resulted in an increase in boiler outlet pressure. This condition in turn reduced the temperature difference between the heating and boiling fluids in the boiler and lowered the heat load as indicated in figure 17. To further illustrate the influence of flow rate and inventory on the operation of the boiler, lines of constant quality have been added to figure 15. It can be seen that both heat load (fig. 17) and vapor quality (fig. 15) were reduced by increasing inventory at a constant flow rate. The system heat load (fig. 17) increased almost linearly with vapor loop flow rate with a fixed inventory but the vapor quality was reduced (fig. 15). The changes in heat load and quality that accompany inventory changes in the boiler are reflected in the condenser and affect the condensing length and condenser inlet pressure which the system seeks for steady-state operation.

Effect of Coolant Inlet Temperature

The effect of coolant temperature on the performance of the vapor loop was determined over a range of vapor loop flow rates at two levels of active inventory and coolant inlet temperatures of 92° and 122° F. As the coolant inlet temperature increases, the temperature differences between the coolant and the condensing vapor decreases, all other independent system variables being held constant. This results in an increase in the length required to complete the condensing process. Changing the distribution of liquid and vapor in the loop was found to change the condenser inlet pressure as indicated in figure 18. Plotted in this figure is condenser inlet pressure against flow rate (similar to fig. 15) for coolant inlet temperatures of 92° and 122° F and two active inventories. The 122° F data were taken directly from figure 15 and those at 92° F were run with fixed inventory and variable vapor loop flow rate. It can be seen that the two sets of curves are of the same general shape but displaced by 2 or 3 psi. The higher pressure occurred with the higher coolant inlet temperature.

The amount of condensate subcooling (saturation temperature minus condenser outlet temperature) is an important parameter in the Rankine cycle since it determines the inlet conditions to the vapor loop pump and the amount of preheating required in the boiler. The effect of vapor loop flow rate and coolant inlet temperature on condensate subcooling is presented in figure 19. The data at 122° F were obtained from the variable inventory data (table II) and cross plotted against flow rate for the one inventory presented. The trend of reducing subcooling with increased flow rate was discussed earlier with reference to figure 10. The data at a coolant inlet temperature of 92° F were obtained with fixed inventory and varying vapor loop flow rate and show the same trend with flow as the 122° F data. As would be expected, the condensate subcooling was reduced by increasing the coolant inlet temperature. Thus the flow rate at which the vapor loop pump cavitated was lower for the higher coolant inlet temperature.

CONCLUDING REMARKS

A Rankine cycle consisting of a liquid heated boiler, a liquid cooled condenser, an unchoked vapor orifice, and a centrifugal pump was operated over a wide range of steady-state conditions. Water was the working fluid. Steady-state operation could be obtained with or without an accumulator between the condenser outlet and the pump.

The effects of vapor loop flow rate and fluid inventory were determined by operating the system over a range of fluid inventories with several fixed values of vapor loop flow rate and fixed inlet conditions (temperature and flow rate) in the heating and cooling loops. For a given vapor loop inventory, it was found that the condenser inventory decreased and the boiler inventory increased as the vapor loop flow rate was increased.

Increases of fluid inventory of the vapor loop with constant vapor loop flow rate resulted in increases in condenser inventory, boiler inventory, and condenser inlet pressure and reductions in system heat load and vapor quality. At a fixed vapor loop flow rate, the sensitivity of the condenser inlet pressure to changes in inventory was found to be much less at the high flow rates than at the lower flow rates investigated. Because this system had no choked station (such as a turbine) to separate the boiler and condenser, the condenser inlet pressure reached by the system for steady-state operation was a function of all of the independent system variables as well as the heat-transfer and pressure loss characteristics of the boiler and condenser. An important parameter in determining the sensitivity of the condenser inlet pressure to changes in inventory was found to be the mean temperature difference between the vapor and the condenser coolant. Where the condenser temperature difference was small (high vapor loop flow rates), the condenser inlet pressure changed only slightly with changes in inventory. Where the temperature difference was high (at lower vapor loop flow rates), however, the condenser inlet pressure was more sensitive to inventory changes.

It was also found that with a fixed inventory the subcooling produced by the condenser decreased with increases in the condenser coolant inlet temperature. The flow rate at which the vapor loop pump cavitated was lower for the higher coolant temperatures.

Lewis Research Center,
National Aeronautics and Space Administration,
Cleveland, Ohio, November 26, 1965.

APPENDIX A

SYMBOLS

C_p specific heat at constant pressure, Btu/(lb)(^{o}F)

h_{fg} latent heat of vaporization, Btu/lb

I active inventory, in.^{3}

I_B boiler inventory, in.^{3}

I_C condenser inventory, in.^{3}

L condensing length, in.

P pressure, psia

Q heat load, Btu/hr

T temperature, ^{o}F

W flow rate, lb/hr

X thermodynamic quality

Subscripts:

B boiler

C condenser

c coolant

H heating

sat saturation conditions

V vapor

1 inlet

2 outlet

APPENDIX B

MEASUREMENT OF LOOP VOLUMES

The total volume of the vapor loop was determined with the loop completely dry. All pressure tap connections, which were filled with liquid during normal operation, were capped and the inventory tank isolation valve was closed. The tank was pressurized with dry nitrogen gas to 100 psia and sealed. The gas in the tank was then allowed to expand into the loop through the isolation valve and the final equilibrium pressure was observed. Sufficient time was allowed for the system to return to room temperature. The volume of the vapor loop was then calculated by using Boyle's law. Repeated measurements agreed within 2 percent. The active volume of the vapor loop was determined by subtracting the volume of the all liquid portion of the loop from the measured total volume of the loop. The volume of the liquid portion of the loop included all the piping from the exit of the condenser tubes to the inlet of the boiler inlet plenum. This volume was measured by the same method as the total loop volume.

APPENDIX C

METHODS OF DATA REDUCTION

System pressures were read from oscillograph records using a visual average of the deflection over a record length of about 40 seconds.

Condensing length was obtained from temperature profiles of the vapor in the two instrumented tubes. The values presented are average values for the two tubes.

The vapor quality was calculated from heat balances written in the boiler and in the condenser:

$$X_{V2B} = \frac{W_H C_{p,H} \Delta T_H - W_V C_{p,V}(T_{sat} - T_{V1B})}{W_V(h_{fg})}$$

Two values of ΔT_H were measured: one by subtracting thermocouple readings at the heating loop inlet and outlet of the boiler, and one by direct reading from a differential thermocouple across the heating side of the boiler.

Similarly,

$$X_{V1C} = \frac{W_c C_{p,c} \Delta T_c - W_V C_{p,V}(T_{sat} - T_{V2C})}{W_V h_{fg}}$$

Again two values of ΔT_c were measured: one by subtracting the coolant inlet from the coolant outlet temperature and one from a differential thermocouple.

The values of quality obtained from these four methods were within 10 percent of each other. Thus the values used herein are the average of the four values obtained. Similarly the heat load is the average of the four methods of calculation.

REFERENCES

1. Zipkin, Morris A.; and Edwards, Russell N., eds.: Power Systems for Space Flight. Vol. 11 of Progress in Astronautics and Aeronautics, Martin Summerfield, ed., Academic Press, 1963.

2. Hooper, J.R.; and Kunz, H.R.: Space Powerplant Startup and Transient Operation Simulation with Direct-Condensing Radiator System. Rep. No. PWA 2342, Pratt and Whitney Aircraft Div., United Aircraft Corp. (NASA CR-54179), 1964.

3. McAdams, William H.: Heat Transmission. Third ed., McGraw-Hill Book Co., Inc., 1954.

4. Carpenter, E.F.; and Colburn, A.P.: The Effect of Vapor Velocity on Condensation Inside Tubes. Proceedings of the General Discussion on Heat Transfer, Inst. Mech. Eng., (London), and ASME, 1951, pp. 20-26.

5. Wyde, S.S.; and Kunz, H.R.: Experimental Condensing - Flow Stability Studies. Rep. No. PWA 2315; Pratt and Whitney Aircraft Div., United Aircraft Corp. (NASA CR-54178), 1964.

TABLE I. - RANGE OF VARIABLES

Heating loop		Cooling loop		Vapor loop	
Boiler inlet temperature, T_{H2B}, ^oF	Flow rate, W_H, lb/hr	Condenser inlet temperature, T_{c1C}, ^oF	Flow rate, W_c, lb/hr	Active inventory, I, $in.^3$	Flow rate, W_V, lb/hr
300	8077	122	6082	Variable (50 to 191)	198
					236
					282
					314
					386
					468
					654
300	8085	92	6060	134	Variable (129 to 664)
	8085	92	6060	99	Variable (129 to 664)

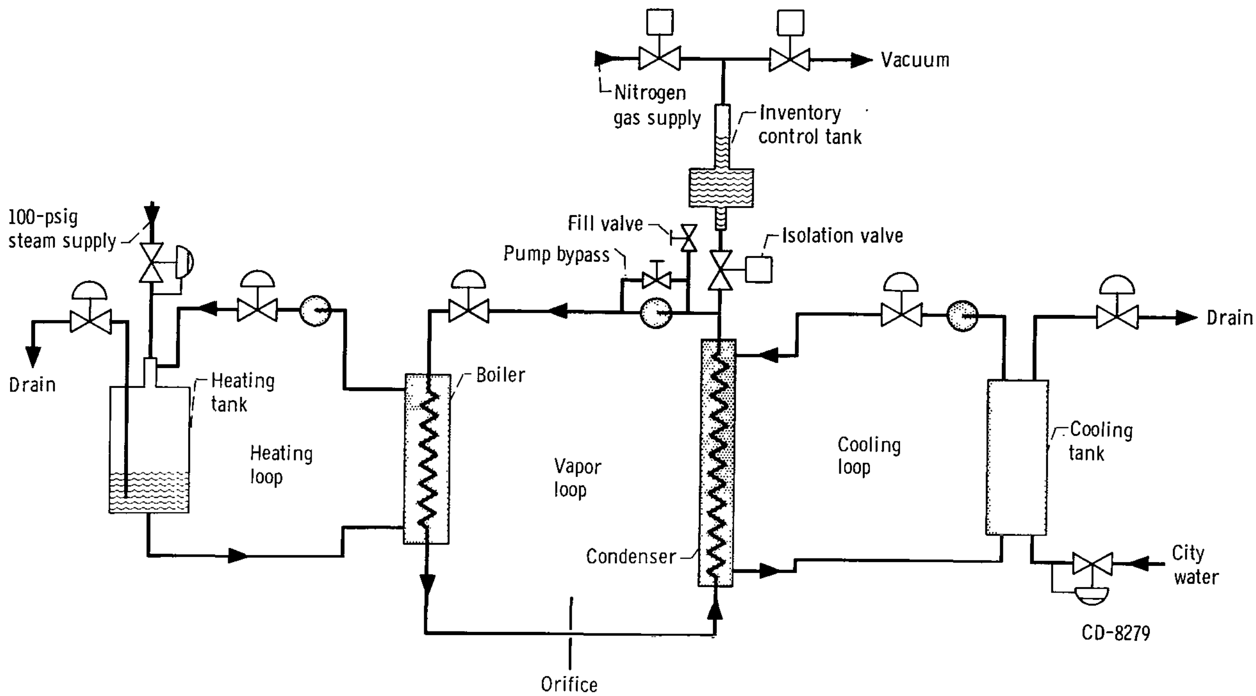
TABLE II. - SYSTEM PERFORMANCE DATA

[Total vapor loop volume, 5.62 gal = 1298 in.^{3}; all liquid volume, 225 in.^{3}.]

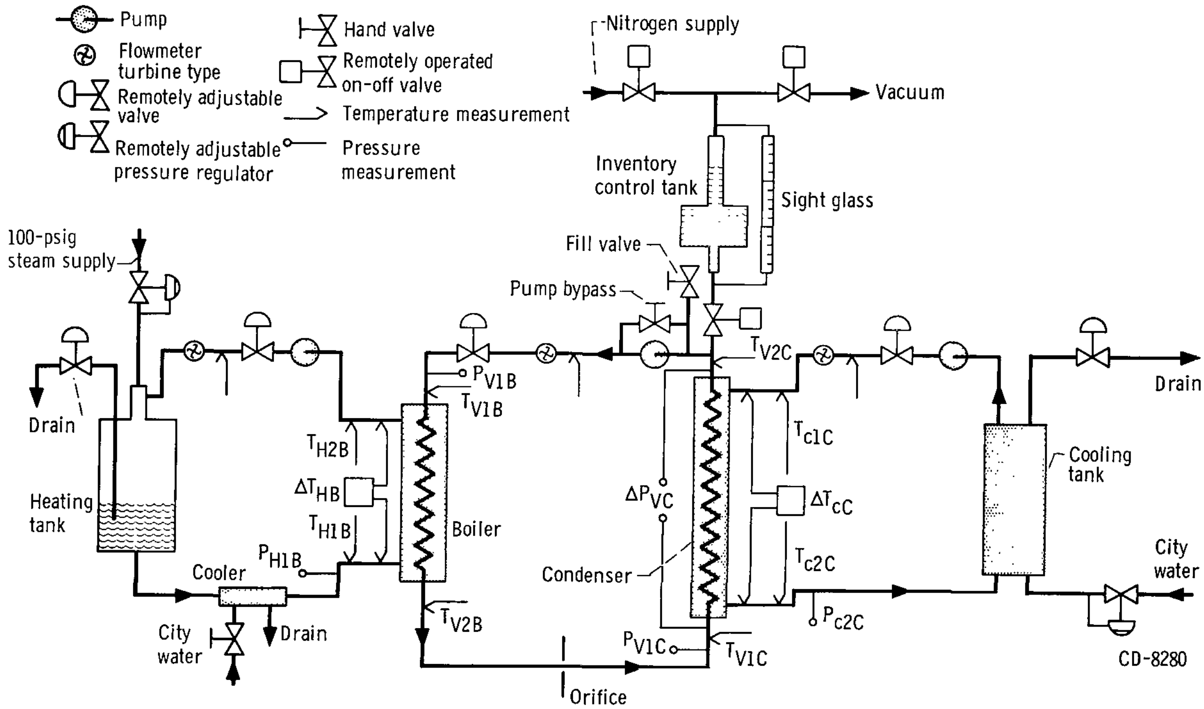
Run	Heating loop				Cooling loop				Vapor loop								Active inventory, I, in.^{3}	Con-densing length, L, in.	Heat load, Q, Btu/hr	Vapor quality, X
	Boiler inlet temper-ature, T_{H1B}, ^oF	Boiler outlet temper-ature, T_{H2B}, ^oF	Boiler temper-ature drop, ΔT_{HB}, ^oF	Flow rate, W_H, lb/hr	Con-denser inlet temper-ature, T_{c1C}, ^oF	Con-denser outlet temper-ature, T_{c2C}, ^oF	Coolant temper-ature rise, ΔT_{cC}, ^oF	Flow rate, W_c, lb/hr	Boiler inlet temper-ature, T_{V1B}, ^oF	Boiler outlet temper-ature, T_{V2B}, ^oF	Con-denser inlet temper-ature, T_{V1C}, ^oF	Con-denser outlet temper-ature, T_{V2C}, ^oF	Flow rate, W_V, lb/hr	Boiler inlet pres-sure, P_{V1B}, psia	Con-denser inlet pres-sure, P_{V1C}, psia	Con-denser pres-sure loss, ΔP_{VC}, psia				
192	300	256	----	8099	122	185	66.2	6037	164	230	228	166	381	21.6	20.1	3.49	87	104.0	38.46×10^4	0.985
193	300	263	----	8069	122	175	54.8	6136	136	241	240	134	385	26.3	25.1	.93	149	52.5	32.30	.771
194	300	272	30.4	8057	121	162	43.1	6038	132	250	250	130	393	30.9	30.1	.21	175	25.0	24.85	.544
195	300	258	43.8	8087	121	180	61.7	6038	142	235	233	140	385	23.5	22.1	1.77	124	67.5	35.75	.875
196	300	254	47.4	8104	120	181	64.4	6188	184	226	222	188	383	20.0	18.4	4.40	55	110 0	38.49	1.0
197	300	256	46.1	8097	122	183	65.2	5987	152	238	236	152	667	25.6	24.0	3.00	156	64.5	37.47	.498
198	300	263	39.2	8047	122	174	55.8	5988	143	245	244	142	660	27.6	27.3	1.00	191	40.0	31.78	.400
199	300	253	49.3	8106	120	186	69.5	5989	174	237	233	176	634	24.6	22.9	5.48	122	91.5	40.30	.600
200	300	252	49.7	8110	120	186	69.9	5989	194	236	233	196	653	24.7	22.9	6.27	108	93.5	40.56	.606
201	301	263	39.6	8094	119	172	55.6	6140	155	218	214	152	309	16.8	15.7	2.64	55	96 0	32 63	1.0
202	300	262	39.6	8098	120	175	58.3	5892	141	226	224	138	314	19.9	19.0	1.65	85	79.0	32 87	.999
203	302	265	38.0	8061	122	174	52.9	6184	135	234	234	132	321	24.2	23.2	.93	115	62.5	31.49	.922
204	302	271	32.2	8036	122	165	45.6	6184	132	246	246	128	309	29.3	28.4	.29	149	39.2	26 59	.786
207	300	273	28.9	8055	122	160	40.8	6285	126	230	230	124	230	22.5	21.1	----	124	44.2	23.92	.976
208	300	272	30.1	8059	122	164	43.4	6135	130	252	252	126	230	16.0	14.8	----	85	64.0	25.02	1.0
209	300	270	30.3	8040	122	166	45.0	6035	137	262	262	134	243	13.4	12.2	----	50	85.5	25.64	.953
210	300	272	27.9	8059	120	162	43.6	5989	128	236	236	122	241	24.9	23.6	----	129	38.5	23.49	.906
211	300	272	28.3	8030	119	158	41.0	6140	127	242	242	121	235	27.7	26.3	----	143	30.5	23.76	.943
212	300	275	26.4	8045	124	160	38.7	6034	129	249	249	126	235	30.9	29.4	----	149	26.2	21.89	.859
213	300	262	39.5	8073	122	175	56.0	6136	134	240	240	130	390	----	----	----	152	48 5	32 71	.769
214	300	262	40.1	8120	123	176	56.8	6035	139	221	221	138	315	18.7	17.3	----	82	81 5	32.75	.994
233	301	259	42.8	8155	121	179	59.8	6137	137	239	237	137	473	24.4	23.8	1.70	154	57.7	35 77	.686
234	300	256	45.4	8167	122	184	63.2	6037	148	235	232	146	468	22.6	22.0	3.03	129	77.5	37.58	.750
235	300	254	47.6	8175	122	188	67.4	6135	165	231	229	167	473	21.4	20.6	4.36	101	99.0	39.88	.811
236	300	254	48.4	8085	121	187	68.1	6038	188	230	226	190	450	20.7	19.8	5.21	80	110 0	39.78	.878
237	300	253	49.3	8108	122	188	69.1	6037	196	230	227	202	477	21.3	20.3	5.64	69	105.0	40.40	.847
238	300	266	36.2	8084	122	171	50.4	6037	136	217	214	132	286	16.1	15.5	1.60	78	77.0	29.57	0.986
239	301	267	34.4	8078	123	172	50.2	6035	132	226	224	128	276	19.1	18.8	.80	101	57.2	29.17	1.0
240	300	268	34.2	8123	121	168	----	6038	129	236	234	125	279	22.9	22.7	.48	124	46.0	27.69	.925
241	301	270	29.5	8110	122	165	----	6036	128	244	244	125	286	26.9	26.7	.32	147	34.2	26.07	.835
242	300	273	28.0	8079	122	158	----	6186	130	249	249	125	282	29.7	29.9	.16	161	23.5	22.47	.715
243	300	278	22.8	8011	121	152	----	6137	128	251	251	122	198	30.2	30.6	.00	147	18.7	18.76	.872
244	300	276	25.2	8015	121	155	36.0	6136	129	235	235	123	198	22.6	22.7	.11	129	31.2	20.38	.987
245	300	277	25.2	7992	121	157	27.6	6185	128	254	255	123	193	17.1	17.0	.32	103	42.5	21.23	1.0
246	300	276	25.2	8020	124	158	----	6182	130	257	264	125	202	13.5	13.4	.58	78	51.2	20 57	.913
247	301	277	25.2	8039	122	159	----	6035	131	242	259	126	198	11 1	10.9	.96	59	73 7	20.91	.965
266	301	266	37.6	8059	93	142	51.5	6074	104	235	234	99	278	23 3	21.2	.48	135	44 2	30.09	.995
267	301	266	36.6	8107	92	140	51.8	6176	104	233	234	98	278	23.3	21.4	.58	134	42.2	30.20	1.0
268	301	269	33.4	8046	92	136	47.2	6175	103	236	236	96	252	22.9	21.2	.42	Inventory	38.0	27.41	.995
270	301	276	26.3	8020	92	128	37.3	6074	102	236	235	94	204	23.9	22.2	.11	tank iso-	28.0	21.40	.959
271	301	280	22.0	8003	92	122	32.2	6075	102	237	236	93	174	23.8	22.4	.05	lation	22.5	18 29	.957
272	301	282	19.5	8041	91	119	29.9	6076	102	237	236	92	156	23.4	22.0	.00	valve	19.2	16.23	.949
273	301	286	16.4	8001	90	113	24.2	6028	104	235	235	91	129	23.2	21.7	.05	closed	13.5	13.97	.999
274	300	263	39.2	8163	90	142	54.2	6077	104	236	235	98	303	23.1	21.4	.74		47.7	31.98	.965
275	301	262	41.4	8145	92	147	58.4	6075	108	235	235	104	330	23.0	21.1	1.01		47.7	33.83	.939
276	301	258	45.0	8112	92	152	62.8	6075	113	236	234	108	392	22.9	21.0	1.44		58.5	36.93	.858
277	302	254	49.3	8104	92	158	69.7	6075	122	234	230	120	472	22.2	20.0	2.77		68.0	40.62	.784
278	300	247	54.2	8081	92	163	75.0	6075	144	230	227	144	664	21.3	18.8	6.17		84.2	44.15	.604
280	300	247	55.1	8179	91	163	76.4	6176	147	231	227	146	656	21.5	18.9	6.33		88.5	45.33	.633
282	300	260	41.4	8102	92	147	57.7	6075	110	235	234	104	352	22.8	21.1	1.01		49.2	33.82	.873
283	300	260	41.5	8128	92	147	58.2	6075	110	235	234	105	357	23.0	21.2	1.12	134	50.7	33.98	.865
285	300	264	37.0	8065	92	143	53.7	6026	104	216	216	100	284	16.2	14.4	.96	101	52.7	30.79	1.0
286	300	264	37.0	8114	92	143	53.7	6025	105	217	216	100	280	16.5	14.6	.96	99	53.2	30.70	1.0
287	300	267	33.8	8052	91	136	48.9	6027	102	215	212	96	252	15.4	13.6	.74	Inventory	50.5	27.78	1.0
288	300	270	29.9	8040	92	134	44.0	6025	102	254	254	96	229	15.1	13.4	.48	tank iso-	45.7	24.98	.971
289	300	274	26.8	8024	93	130	38.7	6074	103	262	262	96	204	14.5	13.0	.43	lation	42.0	22.15	.955
290	300	278	22.9	8007	92	124	34.0	6025	103	256	259	94	174	12.6	11.3	.21	valve	37.5	18 89	953
291	300	281	20.0	8020	92	121	29.4	6024	104	262	253	93	154	11.1	9.8	.16	closed	36.0	17.15	.975
292	300	286	16.0	8028	92	115	25.0	6026	106	255	249	92	129	8.8	7.8	.16		30.2	13 79	.931
293	300	262	39.7	8124	93	148	56.9	6074	109	221	219	104	303	17.5	15.6	1.17		61.0	32.91	1.0
294	300	258	43.9	8089	93	152	61.5	6074	112	224	222	108	330	18.7	16.7	1.54		64.0	35 82	1.0
295	301	254	48.0	8103	92	157	68.3	6025	122	225	224	119	385	19.4	16.9	2.61		74.0	39.62	.962
296	301	250	52.0	8142	91	162	74.1	6026	136	224	224	134	451	19.7	17.3	4.10		87.0	43.03	.900
297	300	246	54.8	8179	93	166	77.3	6074	179	230	227	180	651	21.3	18.3	8.12		101.0	45.30	.672
298	300	257	44.1	8140	92	152	64.2	6075	115	225	221	110	356	19.1	17.0	1.81	99	71.7	36.93	.963

TABLE III. - MAGNITUDE OF OSCILLATIONS WITH FIXED AND VARIABLE INVENTORY OPERATION

Run	Vapor loop				Inventory tank isolation valve position
	Condenser inlet pressure excursion peak to peak, percent of mean	Flow rate excursion peak to peak, percent of mean	Boiler inlet pressure excursion peak to peak, percent of mean	Condenser pressure loss excursion peak to peak percent of mean	
267	6.3	3.8	6.3	73	Closed
268	7.7	4.1	8.6	88	↓
270	7.4	---	7.5	Very large	↓
271	6.7	9.8	8.3	Very large	↓
272	6.1	9.6	6.0	Very large	↓
273	6.2	13.2	3.5	Very large	↓
274	7.0	2.8	6.4	57	↓
275	6.0	2.6	6.8	42	↓
276	7.1	2.2	7.2	44	↓
277	7.5	2.2	8.1	23	↓
278	7.9	6.1	8.1	19	↓
280	9.2	5.6	8.4	18	↓
282	6.4	---	7.2	19	↓
283	4.9	---	5.0	62	↓
192	7.8	7.3	8.4	32	Open
193	6.5	12.3	3.7	53	↓
194	2.0	3.3	2.1	236	↓
195	4.7	3.9	4.6	36	↓
196	6.1	3.9	9.0	24	↓



(a) Basic flow diagram.



(b) Instrumentation.

Figure 1. - Schematic flow and instrumentation diagrams.

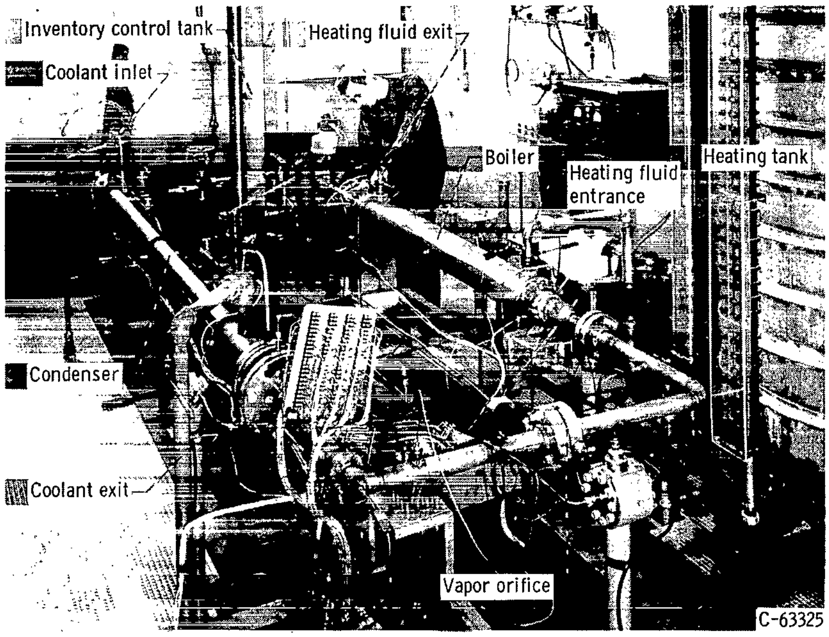


Figure 2. - Water loop installation.

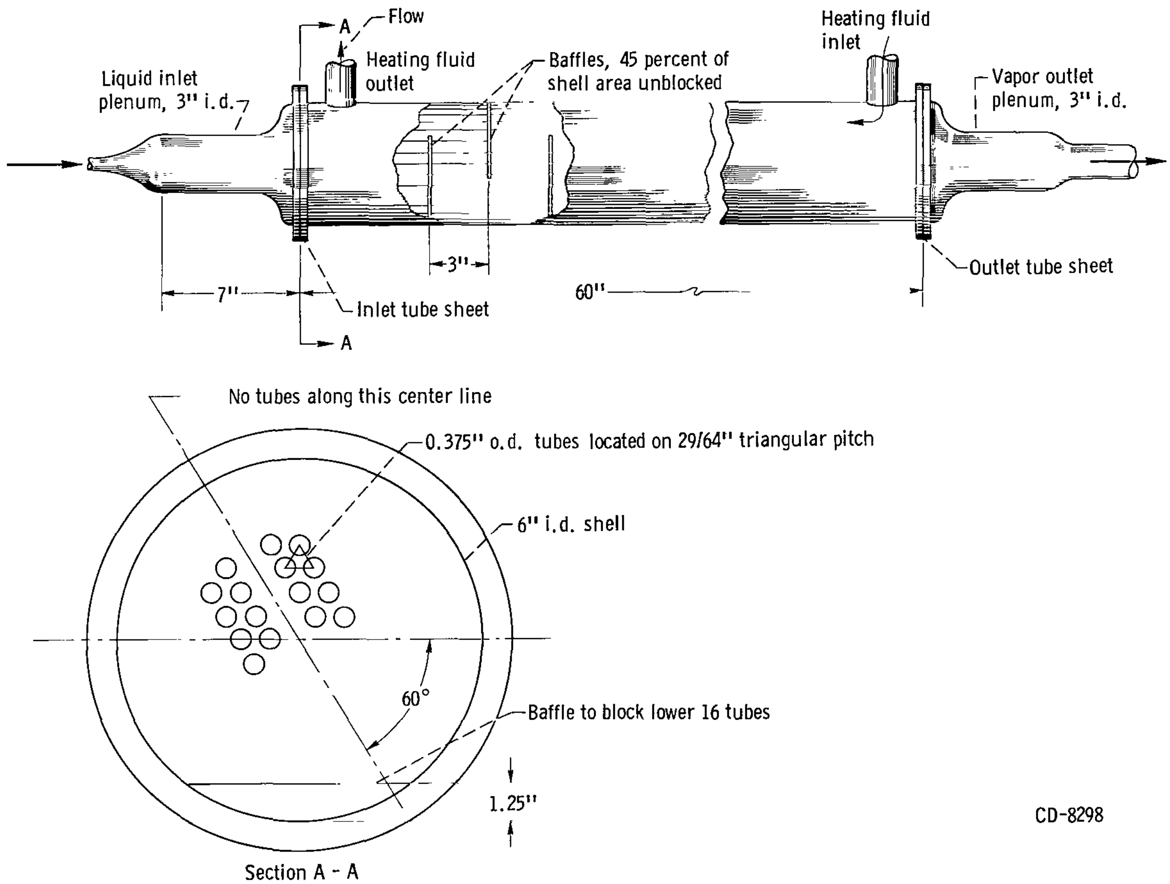
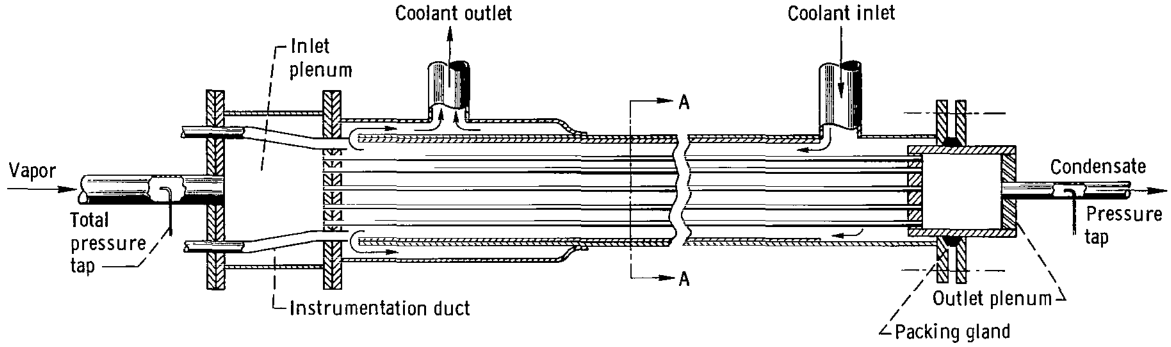
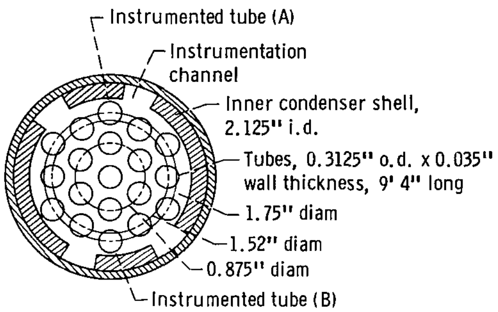


Figure 3. - Boiler schematic.





Section A-A

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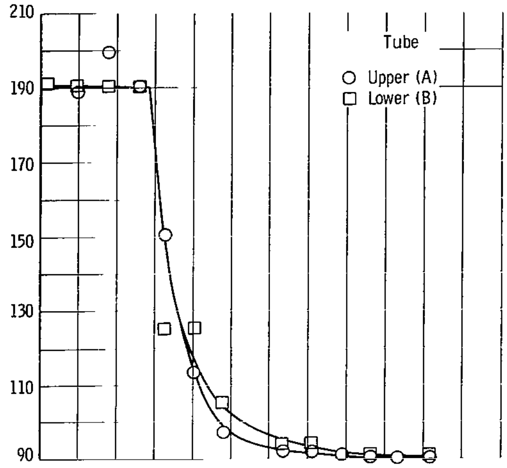
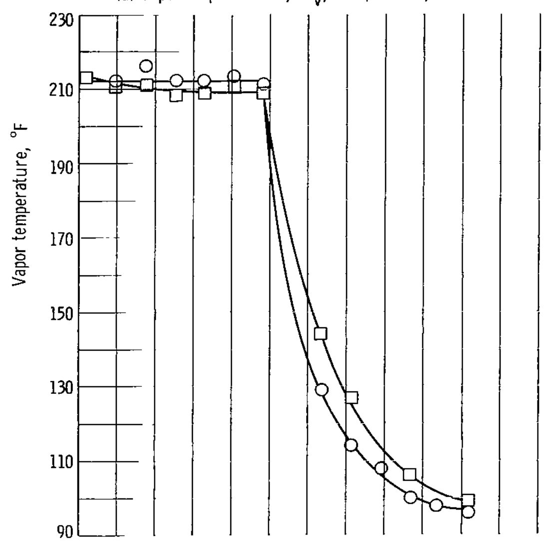
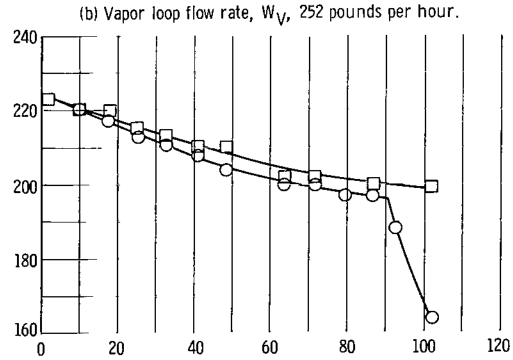
Figure 4. - Sketch of 19 tube condenser.



(a) Vapor loop flow rate, W_V, 129 pounds per hour.



(b) Vapor loop flow rate, W_V, 252 pounds per hour.



(c) Vapor loop flow rate, W_V, 651 pounds per hour.

Figure 5. - Typical vapor temperature profiles in condenser for three vapor loop flow rates. Heating loop boiler inlet temperature, 300° F; heating loop flow rate, 8085 pounds per hour; cooling loop condenser inlet temperature, 92° F; cooling loop flow rate, 6060 pounds per hour; vapor loop active inventory, 99 cubic inches.

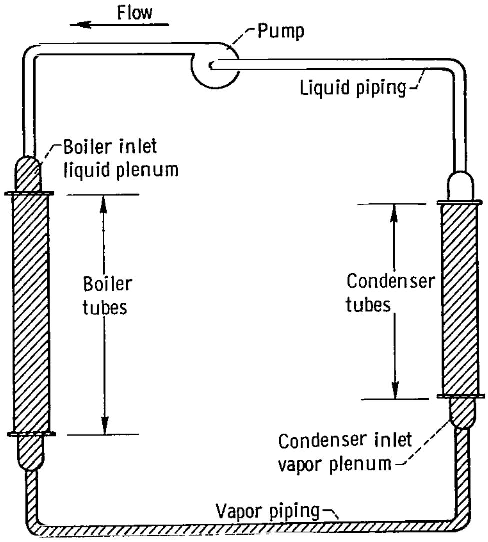


Figure 6. - Definition of active vapor loop volume (shaded area).

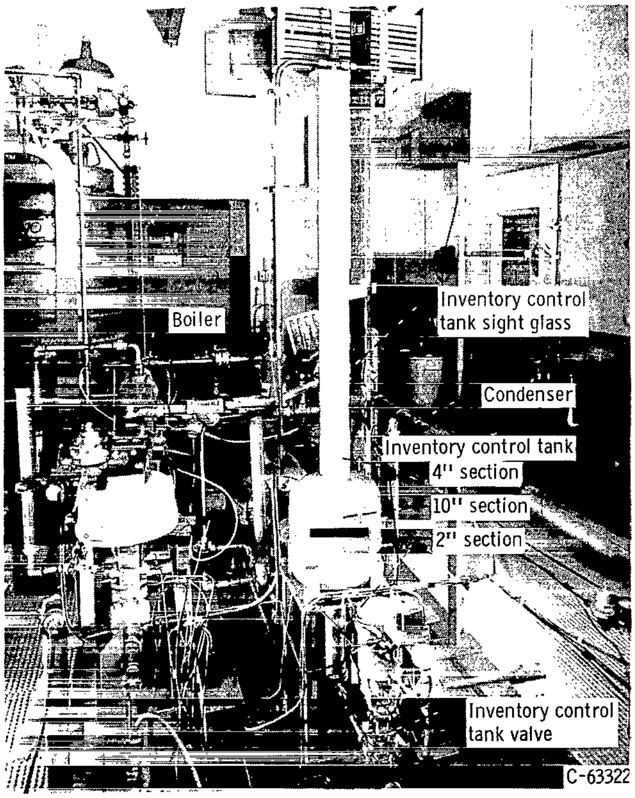
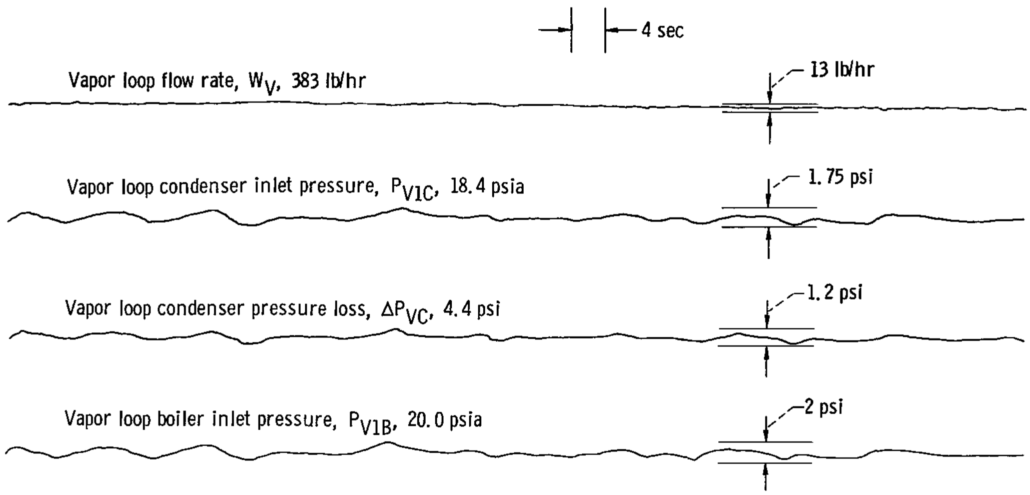
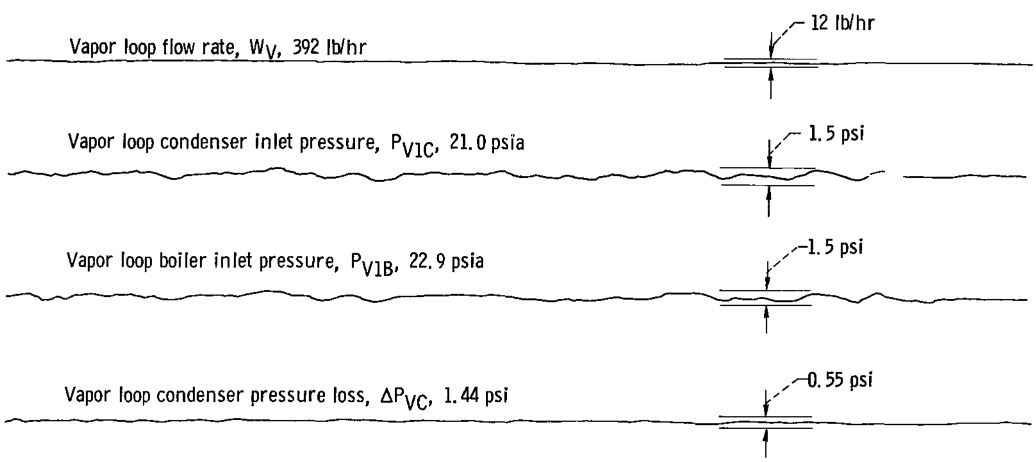


Figure 7. - View of vapor loop showing inventory control tank.



(a) Variable inventory operation. Run 196.



(b) Fixed inventory operation. Run 276.

Figure 8. - Sample steady-state oscillograph recording.

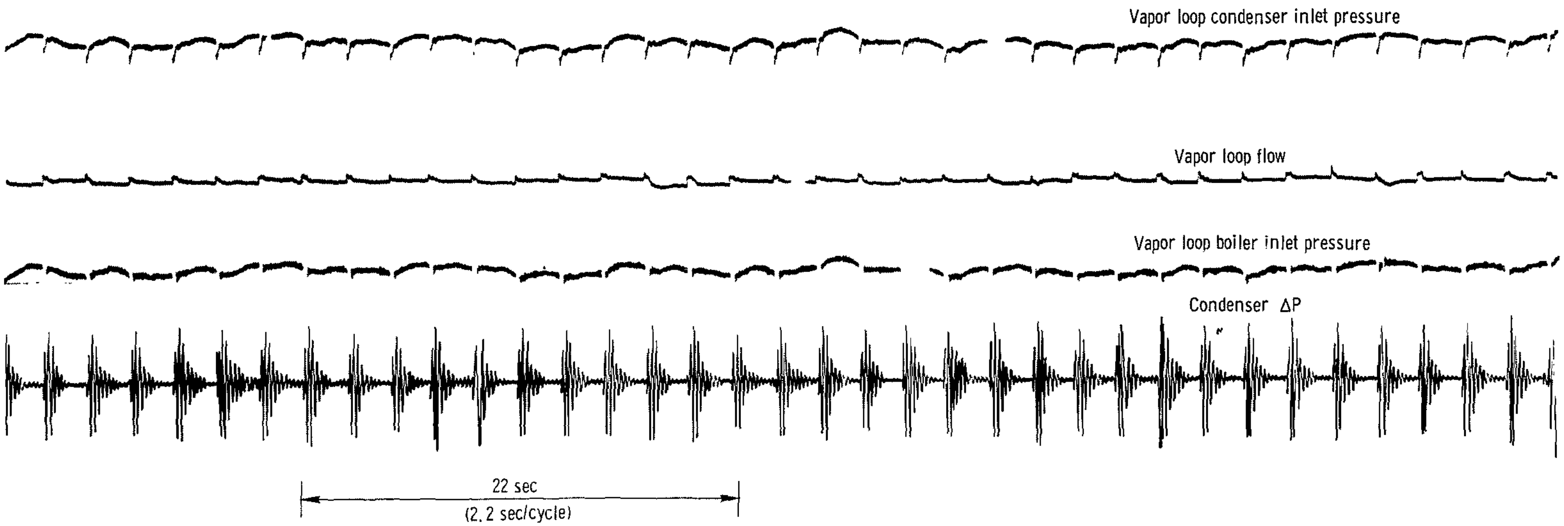
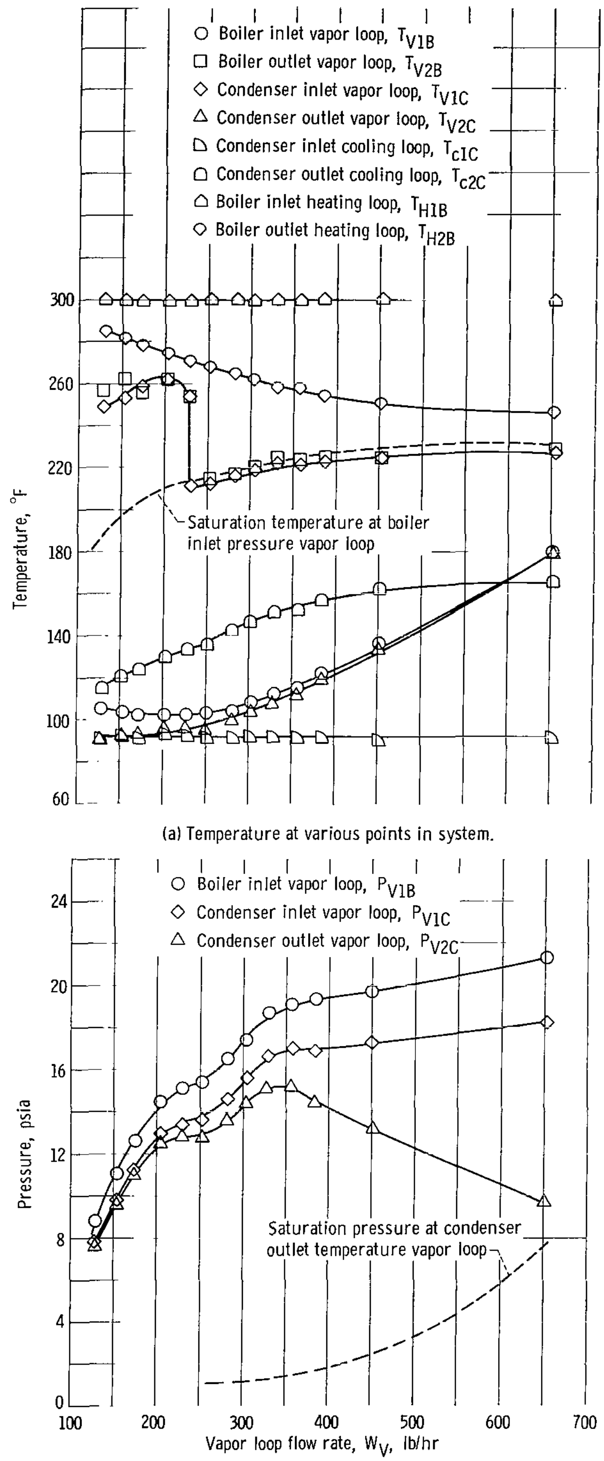


Figure 9. - Typical oscillogram showing slugging in condenser. Heating loop flow, 8001 pounds per hour; cooling loop flow, 6028 pounds per hour; heating loop boiler inlet temperature, 301° F; cooling loop condenser inlet temperature, 90° F; vapor loop flow, 129 pounds per hour; active inventory, 134 cubic inches.



(a) Temperature at various points in system.

(b) Pressures at various points in system.

Figure 10. - Variation of system temperatures and pressures with vapor loop flow rate for active inventory of 99 cubic inches. Heating loop flow, 8079 pounds per hour; cooling loop flow, 6042 pounds per hour.

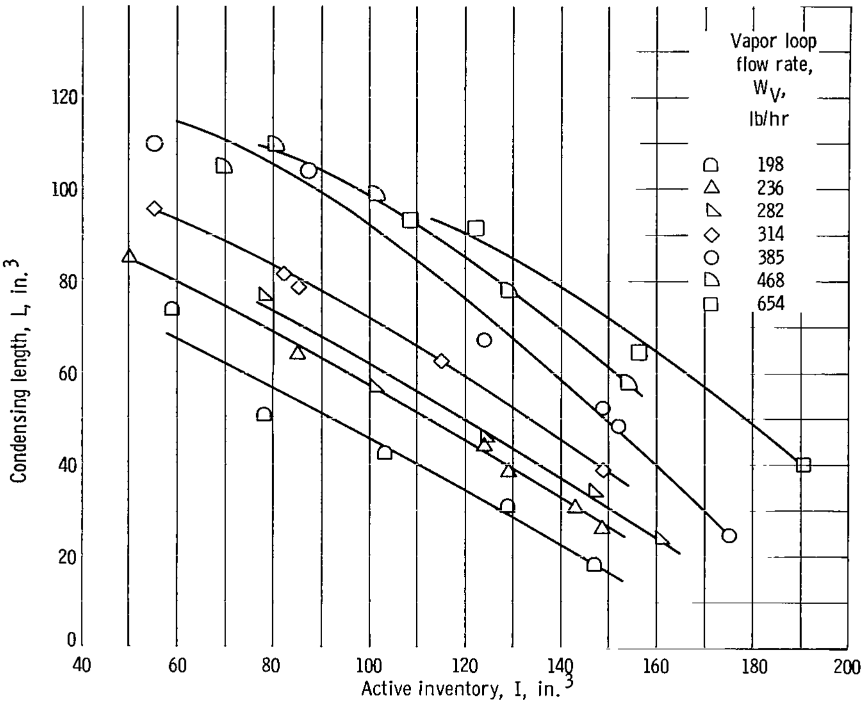


Figure 11. - Variation of condensing length with active inventory for several vapor loop flow rates. Heating loop flow rate, 8077 pounds per hour; heating loop boiler inlet temperature, 300° F; cooling loop flow rate, 6082 pounds per hour; cooling loop condenser inlet temperature, 122° F.

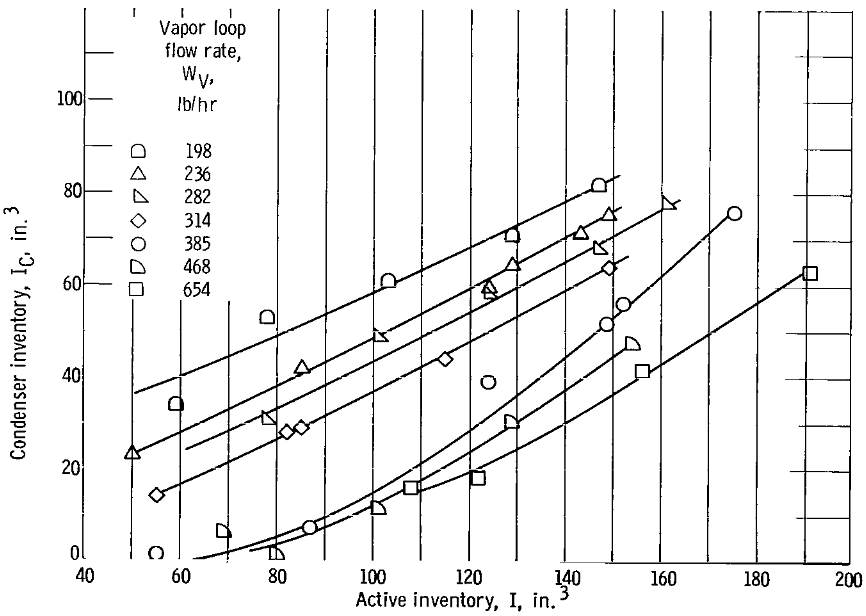


Figure 12. - Variation of condenser inventory with active inventory for several vapor loop flow rates. Heating loop flow rate, 8077 pounds per hour; heating loop boiler inlet temperature, 300° F; cooling loop flow rate, 6082 pounds per hour; cooling loop condenser inlet temperature, 122° F.

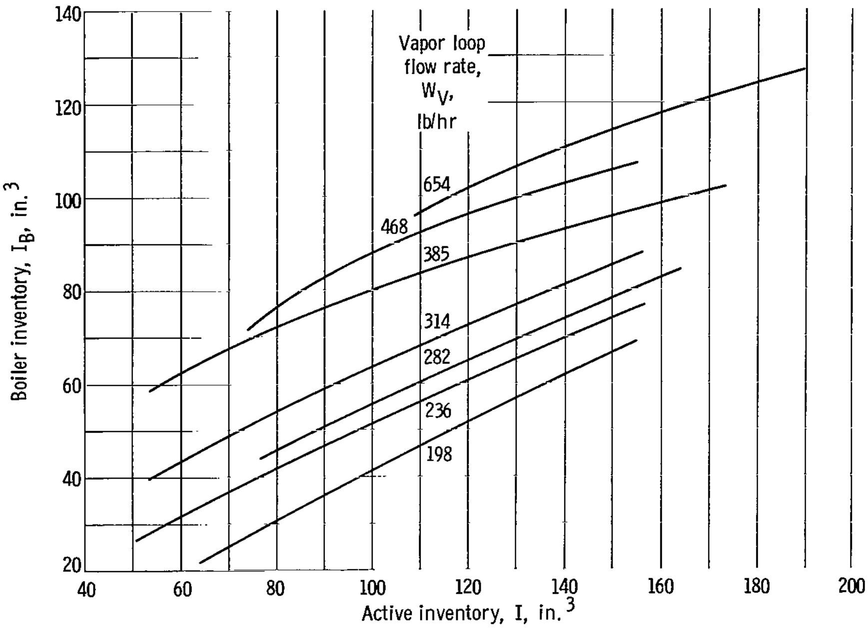


Figure 13. - Variation of boiler inventory with active inventory for several vapor loop flow rates. Heating loop flow rate, 8077 pounds per hour; heating loop boiler inlet temperature, 300° F; cooling loop flow rate, 6082 pounds per hour; cooling loop condenser inlet temperature, 122° F.

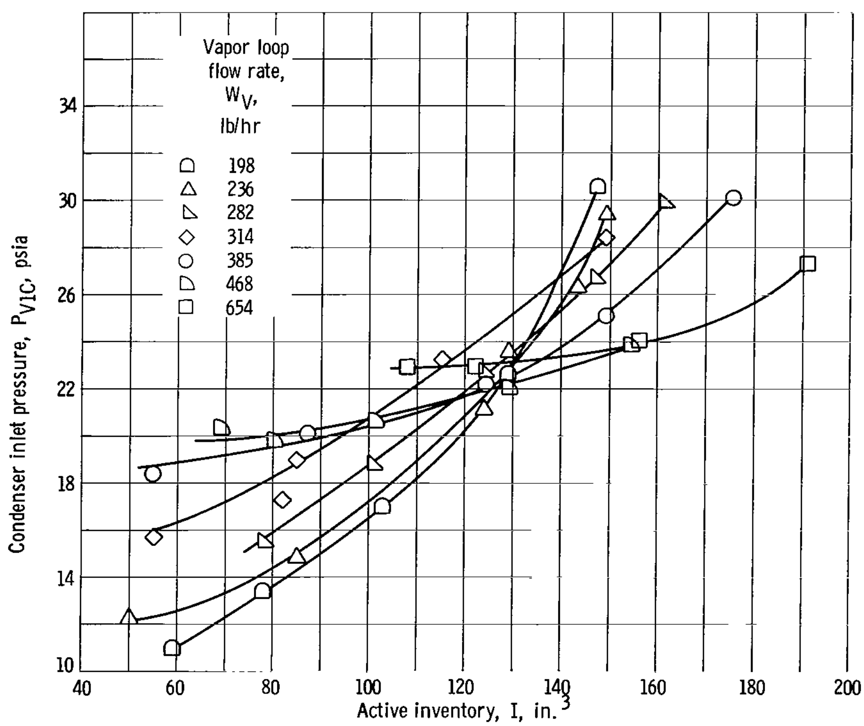


Figure 14. - Variation of vapor loop condenser inlet pressure with active inventory and vapor loop flow rate. Heating loop flow rate, 8077 pounds per hour; heating loop boiler inlet temperature, 300° F; cooling loop flow rate, 6082 pounds per hour; cooling loop condenser inlet temperature, 122° F.

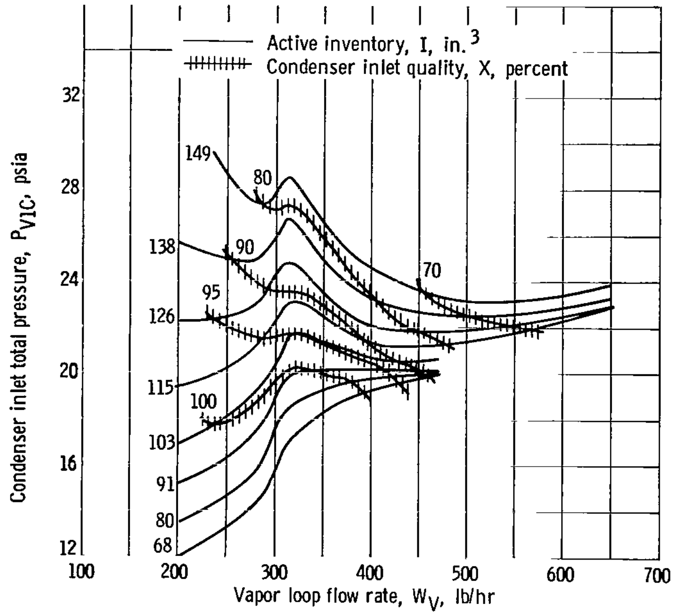


Figure 15. - Condenser inlet pressure against vapor loop flow for various vapor loop inventories. Heating loop flow rate, 8077 pounds per hour; heating loop boiler inlet temperature, 300° F; cooling loop flow rate, 6082 pounds per hour; cooling loop condenser inlet temperature, 122° F.

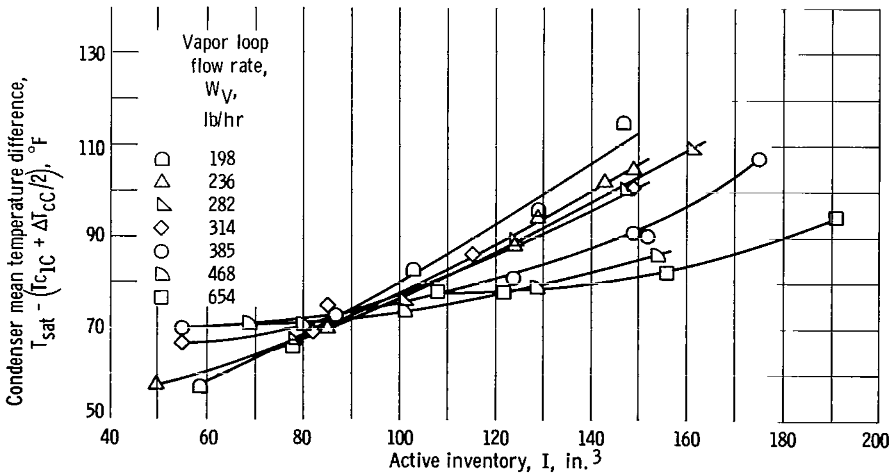


Figure 16. - Variation of temperature difference between vapor and condenser coolant with active inventory for several vapor loop flow rates. Heating loop flow rate, 8077 pounds per hour; heating loop boiler inlet temperature, 300° F; cooling loop flow rate, 6082 pounds per hour; cooling loop condenser inlet temperature, 122° F.

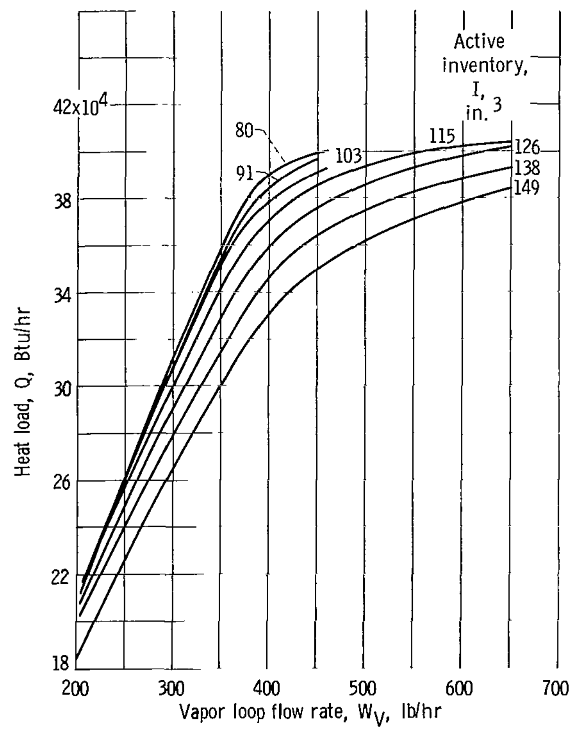


Figure 17. - Variation of heat load with vapor loop flow rate for several active inventories. Heating loop flow rate, 8077 pounds per hour; heating loop boiler inlet temperature, 300° F; cooling loop flow rate, 6082 pounds per hour; cooling loop condenser inlet temperature, 122° F.

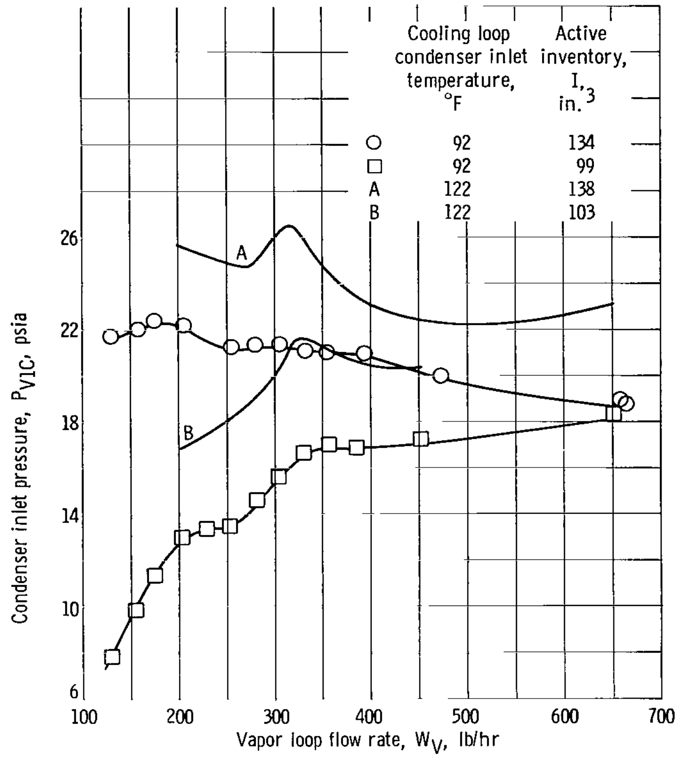


Figure 18. - Effect of coolant inlet temperature on vapor loop pressure at condenser inlet. Heating loop flow rate, 8080 pounds per hour; heating loop boiler inlet temperature, 300° F; cooling loop flow rate, 6072 pounds per hour.

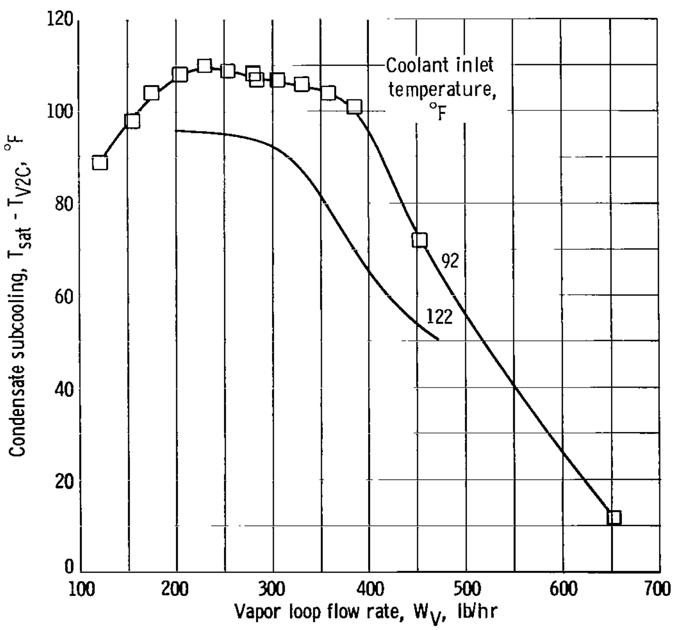


Figure 19. - Effect of coolant inlet temperature on subcooling at condenser outlet. Heating loop flow rate, 8080 pounds per hour; heating loop boiler inlet temperature, 300° F; coating loop flow rate, 6072 pounds per hour; active inventory, 99 cubic inches.

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